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Heat Transfer Between Gas and Cylinder Wall of Refrigerating Reciprocating Compressor

R. Liu

Z. Zhou

HEAT TRANSFER BETWEEN GAS AND CYLINDER WALL OF REFRIGERATING RECIPROCATING COMPRESSOR

Ruihu Liu
Research Assistant

Zicheng Zhou
Associate Professor

Xian Jiaotong University
The People's Republic of China

ABSTRACT

An investigation was made to determine the temperature distribution on the cylinder wall of a reciprocating refrigeration compressor and the coefficient of heat transfer between the cylinder wall and the gaseous refrigerant. The purpose was to study the working process of a refrigerating compressor and to find correlations that can be applied in computer modeling of this type of compressor.

To determine the coefficient of heat transfer and the temperature distribution on the cylinder wall theoretically is rather difficult. Usually they are determined experimentally.

The temperature distribution on the cylinder wall was measured at different pressure ratios,different suction temperatures and different speeds. The correlations are obtained by using the least square method.

The heat flow rate can be calculated based on the differential equation for the first law of thermodynamics, the data of P-φ diagram and the data of valve displacement using the method of orthogonal polynomial fitting.

Investigation has shown that the temperature distribution on the cylinder wall varies with the pressure ratio,suction temperature,speed,oil temperature and with location. Correlations were developed.

INTRODUCTION

Over the years,computer simulation of refrigeration reciprocating compressor has developed rapidly,and is becoming more and more powerful tool for analysing and improving the characteristics of refrigerating reciprocating compressors. Most compressor modeling desingers pay attention to the valve moving modeling,but pay little attention to the heat transfer modeling. Of course, a correct valve behavior is necessary for a good compressor modeling,but it is also important for the heat transfer. A correct heat transfer behavior not only influences the compression and re-expansion process,but also influences the suction and discharge process.

From the thermodynamic point of view,heat transfer from the gas to the surrounding during the working processes will reduce the compression work. Therefore,the EER of compressor will be improved.

It is our direct aim to determine correlations which discribe the temperature distribution on the cylinder wall and the heat flux through the cylinder wall for a certain type of compressor. But our indirect and more significant aim is to use our reciprocating refrigerating compressor modeling, which is developed by using our heat transfer correlation,to do other investigations. The work described in this paper is the first step of our project.

On each fixed point of the cylinder wall, temperature is variable with time. Experiment[1] shows that they are only variable within 1 °C. Owing to the thermal inertia of the cylinder material, its temperature variation is smaller then the gaseous temperature variation in the cylinder. So,temperature gradients on the cylinder wall, in our case,are taken as the average values during one cycle.

The experimental compressor is a Chinese made R12 high speed refrigeration compressor. It has two cylinders, 100 MM bore and 70 MM stroke. The crank shaft speed is 1440 R.P.M. .

EQUATIONS USED TO DETERMINE THE HEAT FLOW RATE

Let us analyse a control volume shown in Fig. 1. According to the first law of thermodynamics

$$\dot{Q}=\frac{\partial U}{\partial t}+\sum_{out}\dot{m}h-\sum_{in}\dot{m}h+\dot{W} \quad (1)$$

using the ideal gas equations and negelecting the gas leakage,we get:
during the compression and re-expansion process

$$\frac{dQ}{d\varphi}=\frac{1}{(k-1)}\frac{d(pV)}{d\varphi}+p\frac{dV}{d\varphi} \quad (2)$$

during the suction process

$$\frac{dQ}{d\varphi}=\frac{1}{(k-1)}\frac{d(pV)}{d\varphi}+p\frac{dV}{d\varphi}-\frac{dm_s}{d\varphi}C_pT_s \quad (3)$$

during the discharge process

$$\frac{dQ}{d\varphi}=\frac{1}{(k-1)}\frac{d(pV)}{d\varphi}+p\frac{dV}{d\varphi}+\frac{dm_d}{d\varphi}C_pT_d \quad (4)$$

To solve the equations (2)-(4),we need to know the values of p,V,$\dot{m}_s$,$\dot{m}_d$,T_s,and T_d. P, V,T_s and $T_d$can be measured directly, but the instantaneous $\dot{m}_s$ and $\dot{m}_d$ are difficult to be measured accurately. It can be calculated by using the valve equations if the valve displacement,coefficient of flow,and coefficient of force have been measured.

The mass flow rate through the suction valve is

$$\frac{dm_s}{d\varphi}=m_{ss}\mathrm{EXP}\left[\int_{\varphi_{ss}}^{\varphi}f(\varphi)d\varphi\right]\cdot f(\varphi) \quad (5)$$

where

$$f(\varphi)=\frac{[\alpha A]_s}{\omega V}\sqrt{\frac{2kRT_s}{k-1}\left(1-\left(\frac{p}{p_s}\right)^{\frac{k-1}{k}}\right)} \quad (6)$$

also for the discharge valve

$$\frac{dm_d}{d\varphi}=G(\varphi)/\left[m_{ds}^{3/2}+\frac{3}{2}\int_{\varphi_{ds}}^{\varphi}G(\varphi)d\varphi\right]^{1/3} \quad (7)$$

where

$$G(\varphi)=p_d\frac{[\alpha A]_d}{\omega RT_d}\sqrt{\frac{2k}{k-1}}\sqrt{pV\left[1-\left(\frac{p_d}{p}\right)^{\frac{k-1}{k}}\right]} \quad (8)$$

the gas volume in the cylinder is

$$V=V_0+\frac{\pi d^2r}{4}\left[(1-\cos\varphi)+\frac{\lambda}{4}(1-\cos2\varphi)\right] \quad (9)$$

C_p can be calculated by

$$C_p=\sum_{i=0}^{m}a_iT^i \quad (10)$$

EXPERIMENTAL SCHEME

In order to measure the temperature distribution on the cylinder wall, cylinder head, gas temperature in the suction plenum and lubricating oil temperature,about twenty copper-constantan thermocouples were arranged at different locations,to sense the temperatures. Several thermocouples were located along the axis,others around cylinder circumference.

A potentiometer was used to measure the voltages across the thermocouples. Fig. 2 shows the temperature measuring scheme.

Fig.3 shows the measuring scheme by which the indicated diagram and the suction and discharge valve displacement were measured. A pressure transducer and valve displacement transducer were located in the cylinder head. A dynamic strain apparatus was used to amplify the signals. A light-oscilloscope was used to photograph the p-φ diagram. An electric-oscilloscop,a magnetic recorder and a data analyzer were used to get the pressure-crank angle numerical data.

TEMPERATURE GRADIENT ON THE CYLINDER WALL AND ITS INFLUENTIAL FACTORS

Experiments show that the temperature distributions are variable with different locations,pressure ratios,speeds,suction temperatures and lubricating oil temperatures. The influencing factors are as follow:

a) Pressure ratio

Pressure ratio influence on temperature gradient of wall is obvious. The discharge temperature of the gas will be increased when the pressure ratio increases. therefore,the temperature of wall will rise respectively. Fig.4 shows the temperature distribution of the wall along the axis. Why does temperature decrease at the position near x=20-40 MM? The answer is that the outside of cylinder wall was cooled by suction gas. Fig.5 shows temperature distribution around the cylinder circumference on two different sections. Fig.5a is the section at x=61 MM, at a certain pressure ratio, the location of high temperature is located near 90 degree which is close to the discharge pipe. The lower temperature is near 270 degree which is close to the suction plenum. Fig.5b is the section at x=110 MM. It can be seen that the temperature distribution is different from Fig.5a.At a certain pressure ratio, the highest temperature position is near 270 degree.This is because of friction.During the piston moving,it has friction against the cylinder wall at that side. The heat which is generated by frictional energy conducts to the wall. Fig.6 shows the cylinder head temperature and the gas temperature in the suction plenum.

b) Speed

When the speed increases, the compressor capacity increases. Therefore,the temperature of the wall rises. See Fig.7.

c) Suction temperature

The temperature distribution on the cylinder wall is nearly a linear function of suction temperature. Fig.8 shows that the temperature on the wall and on the head vary with suction temperature. The temperature of gas in the suction plenum is also a linear function of suction temperature.

d) Temperature of oil

Experiment shows that the oil temperature influences mainly to the end part of the wall,which is near to the crankcase. This can also be seen from Fig.4. In that part area,the cylinder is contacted with the gas and oil in the crankcase instead of the gas in the suction plenum. Several factors cause the oil temperature to vary.Two of them are the speed and the suction temperature,others are the cooling effect of the lubricating oil,the crankcase surface area and the amount of oil in the crankcase. Fig.9 shows the oil temperature versus speed and suction temperature.

HEAT FLOW RATE

Heat flow rate can be calculated by using equations(2)-(10). the p-φ data picked up from the experiment are the average values during 64 cycles. A curve shown in Fig.10 was fitted by using the orthogonal polynomial fitting method. It can be seen that heat is rejected from the gas to the cylinder during the half crank angle of the re-expansion process,and about three fifth crank angle during the compression process. Heat is rejected during the whole discharge process,and is absorbed during the whole suction process.

DATA CORRELATIONS

The temperature of cylinder wall

$$t_{ww}=24.32+0.7191t_s+5.64\varepsilon-17.936\bar{s}+14.183\bar{s}^2 \quad {}^\circ C \quad (11)$$

equation (11)is limited to use for $\bar{s}\leqslant 1$.
The temperature of cylinder head

$$t_{wc}=13.64+0.1791t_s+11.235\varepsilon \quad {}^\circ C \quad (12)$$

The temperature of gas in the suction plenum

$$t_{sr}=12.4806+0.8418t_s+3.323\varepsilon+0.00369n \quad {}^\circ C \quad (13)$$

The temperature of the lubricating oil

$$t_{oil}=13.52-0.05n-0.000159n^2 \quad {}^\circ C \quad (14)$$

A correlation of the form

$$Nu=Const\ Re^n Pr^m \quad (15)$$

would be used to evaluate the heat transfer rate,we got

Const=0.75 , n=0.8 ; m=0.6 .

The swirl-squish velocity, Reynolds number are defined as follow:

Swirl-squish velocity

$$\omega_g=\begin{cases} 2\omega(1.04+0.45\cos 2\varphi), & \frac{\pi}{2}<\varphi<\frac{3}{2}\pi \\ \omega(1.04+0.5\cos 2\varphi), & \frac{3}{2}\pi<\varphi<\frac{\pi}{2} \end{cases} \quad (16)$$

Reynolds number

$$Re=\frac{\rho De^2\omega_g}{2\mu} \quad (17)$$

equvalent dimeter

$$De=\frac{3d\cdot s}{2s+d} \quad (18)$$

CONCLUSION

The temperatures on the cylinder wall are variable and are functions of the pressure ratio,speed,suction temperature and the location. Correlations can be used in the compressor modeling.

A heat flow rate fuction has been obtained. It can be used in compressor modeling also.

Further investigation will be done with different refrigerants and different sizes of compressors.

NOMENCLATURE

A - area M^2
a - coefficient in Eq.(10)
c - gas velocity M/S
d - cylinder bore M
h - enthalpy J/Kg
$\dot{m}$ - mass flow rate Kg/s
k - ratio of specific heats
p - pressure N/M^2
$\dot{Q}$ - heat flow rate W
R - gas constant J/Kg °C
r - crank radius M
s - piston stroke M
T - absolute temperature K
t - temperature °C or time S
U - internal energy J
V - volume M^3
$\dot{w}$ - work transfer rate J/S
x - distance from the cylinder head to a certain point of wall MM
$\bar{s}$ - x/d (here d in MM)
Nu - Nusselt number
Pr - Prandlt number
Re - Reynolds number
α - coefficient of flow through the valve
ω - angular velocity of crank shaft Deg/S
ρ - density Kg/M^3
μ - visconsity M^2/S
ε - pressure ratio

SUBSCRIPTS

s - suction
d - discharge
in - flow into the cylinder
out - flow out of the cylinder
ss - at the begining of the suction process
ds - at the begining of the discharge process
ww - cylinder wall
wc - cylinder head
sr - suction plenum
oil - oil

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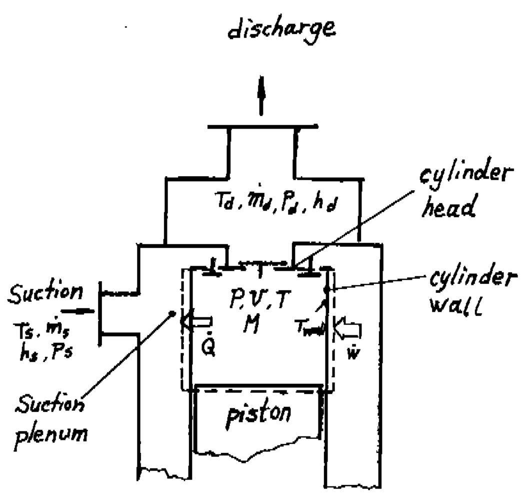

Fig.1 The Control Volume

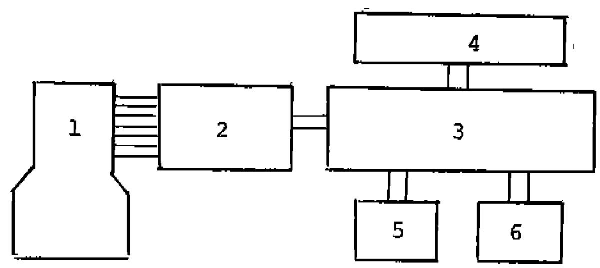

Fig.2 The Temperature Measuring Scheme

1.- Compressor
2.- Switch
3.- UJ-35 Potentiometer
4.- AC/5-15 Microamp
5.- Standard Battery
6.- Power

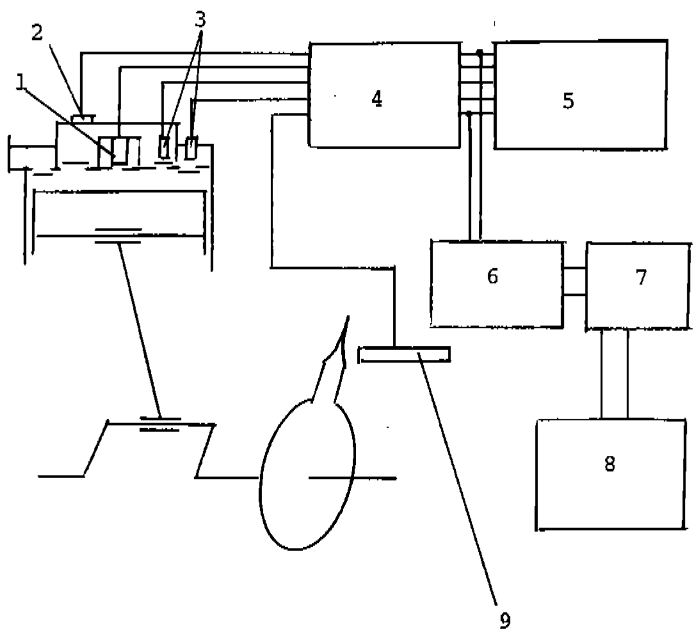

Fig.3 The p-φ Diagram,Valve Displacement and The Numerical Data Measuring Scheme

1.- Cylinder Pressure Transducer
2.- Discharge Plenum Pressure Transducer
3.- Valve Displacement Transducer
4.- Y6D-3A Dynamic Strain Apparatus
5.- Light-Oscilloscop
6.- Magnetic Tape Recorder
7.- Electrical-Oscilloscope
8.- 7TZ08 Data Analyzer
9.- Dead Centre Transducer

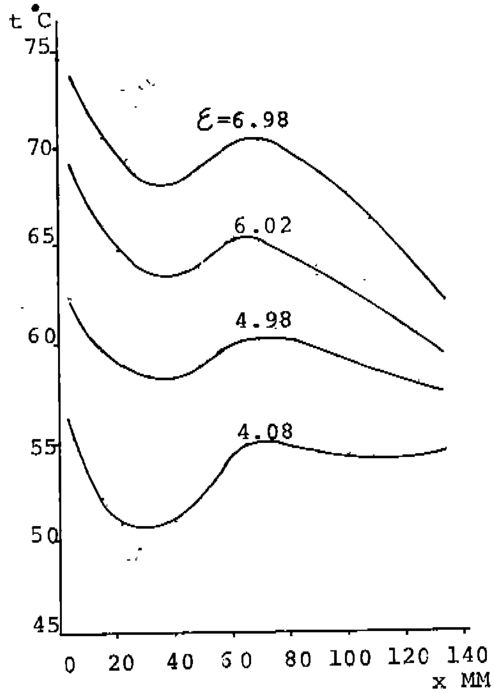

Fig.4 Temperature Distribution of Cylinder Wall Along the Axis

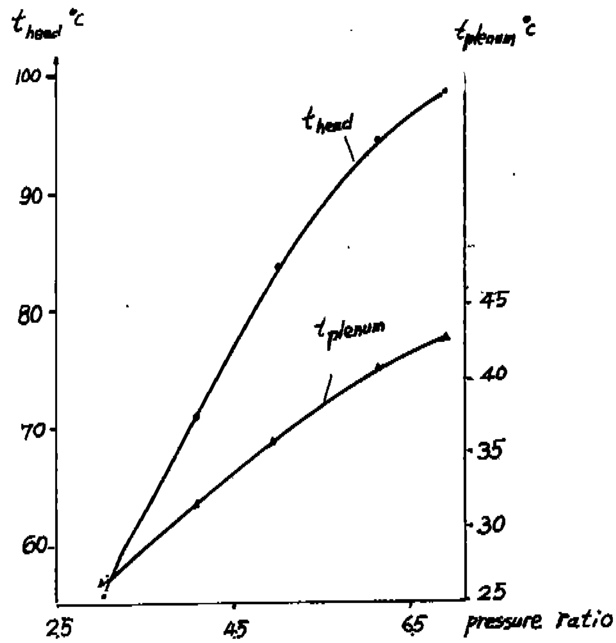

Fig.6 The Cylinder Head Temperature and The Gas Temperature in the Suction Plenum

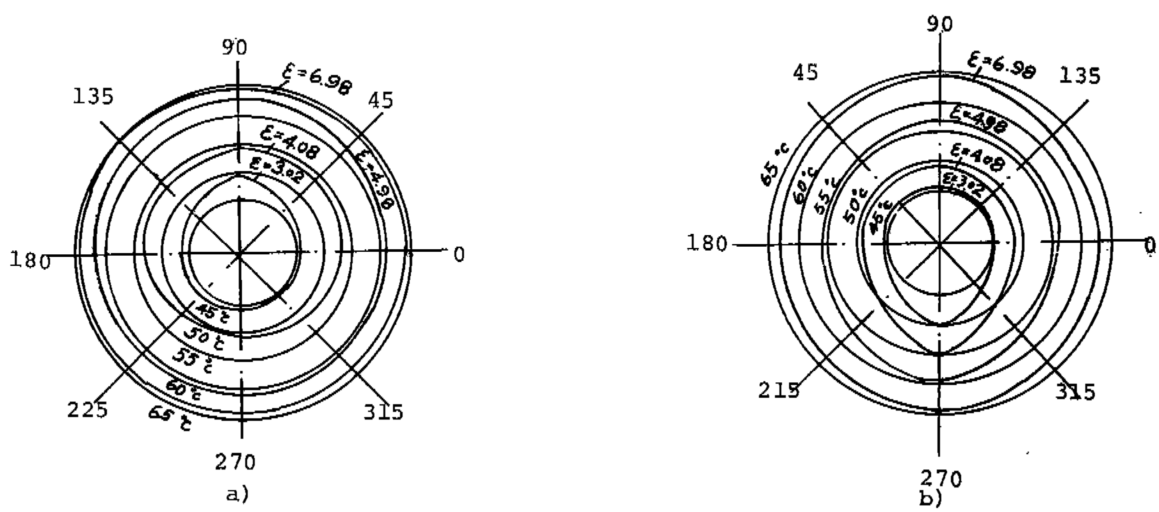

Fig.5. Temperature Distribution of Cylinder Wall Around the Circumference

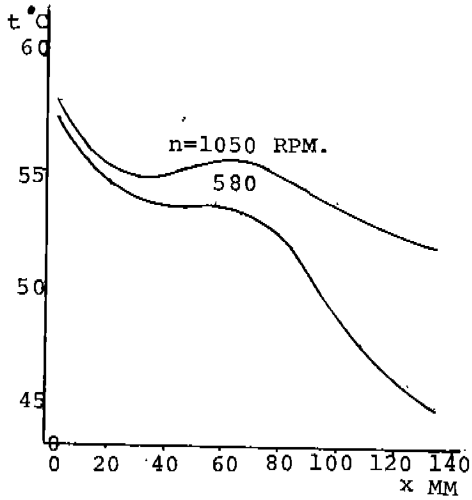

Fig.7 The Temperature of The Cylinder Wall Along the Axis

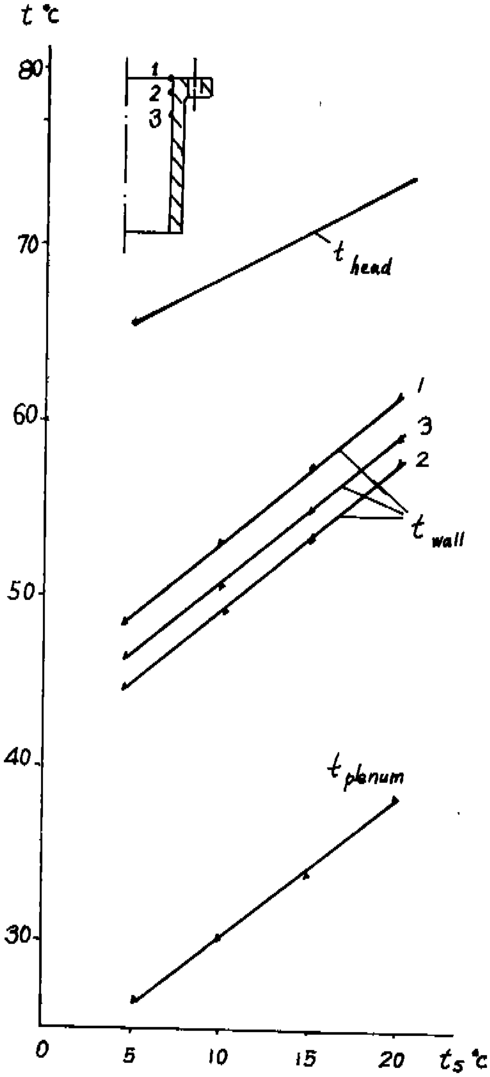

Fig.8 The Temperatures On the Cylinder Wall, On the Cylinder Head and the Gas Temperature in the Suction Plenum

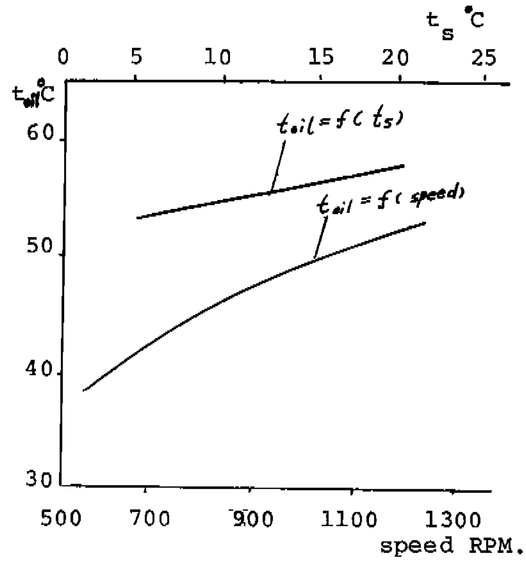

Fig.9 The Temperature of Oil Versus the Speed and the Suction Temperature

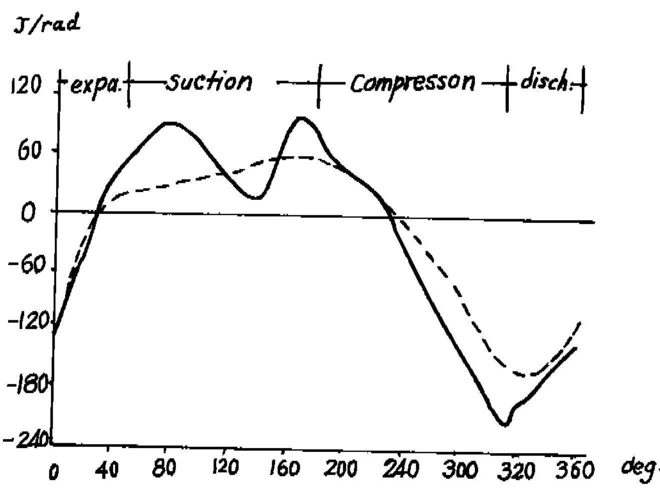

Fig.10 The Heat Flow Rate

—— Experimental Value
---- Calculation Value According to the Eq.(15)